RAPID EMPLACEMENT AND RETRIEVING DEVICE FOR GROUND STAKES GP-112/G AND GP-113/G

REPORT NO. 3 THIRD QUARTERLY REPORT
1 JANUARY 1963 TO 31 MARCH 1963

CONTRACT NO. DA-36-039 SC-90760

U. S. ARMY ELECTRONICS RESEARCH AND DEVELOPMENT LABORATORY
FORT MONMOUTH, NEW JERSEY

AIRCRAFT ARMAMENTS, Inc.
COCKEYSVILLE, MARYLAND

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Report No: ER-2843B

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I. PURPOSE

The purpose of this contract is to provide a comprehensive feasibility study, evolve a design plan, and develop a ground stake emplacement-retrieving device. Capabilities of the device should include emplacement and retrieving of the GP-112/G and GP-113/G Ground Stakes within one minute in various specified soils (1-1/2 minutes in darkness). Other objectives are weight, 25 pounds; length, 42 inches; and to be non-expendable.

The work conducted during the third quarter has included: preparation of a design plan, preparation of visualization data, preparation of detail drawings, and experimental emplacement and retrieving employing breadboard components.

II. ABSTRACT

Experimentation has been conducted to assess the energy required to emplace Ground Stakes GP-112/G and GP-113/G under typical and extreme soil conditions. Results of these tests, as well as prior tests and analytical investigation, were incorporated into the design and presented in a Design Plan. Visualization data was prepared and delivered. Detail drawings have been prepared and a stress analysis of all parts completed. Analysis and experimentation have been conducted to evaluate various retrieve approaches.

Following approval of the Design Plan by USAELRDL, drawings were released for the fabrication of the first of three feasibility models of the ballistic impact hammer.

The breadboard model of the ballistic impact hammer, employed in preliminary tests, was successfully demonstrated ~~for USAELRDL personnel~~ on 25 ~~February 1963 at Fort~~ Monmouth~~, New Jersey.~~

III. PUBLICATIONS, REPORTS & CONFERENCES

During the period covered by this report, two conferences occurred between representatives of USAELRDL and AAI. On 30 January 1963 USAELRDL was visited by R. G. Strickland and A. C. Powell to review the Design Plan and deliver visualization data. It was agreed at this conference that USAELRDL would accept the Plan but AAI would cover three areas in greater detail and resubmit the plan. The areas to be expanded upon were: retrieve approach, tool roll carrier, and test schedule. A second trip to USAELRDL was made on 25 February 1963 by R. G. Strickland, N. J. LaCosta and A. C. Powell. A presentation and demonstration of the ballistic impact device was made for USAELRDL personnel.

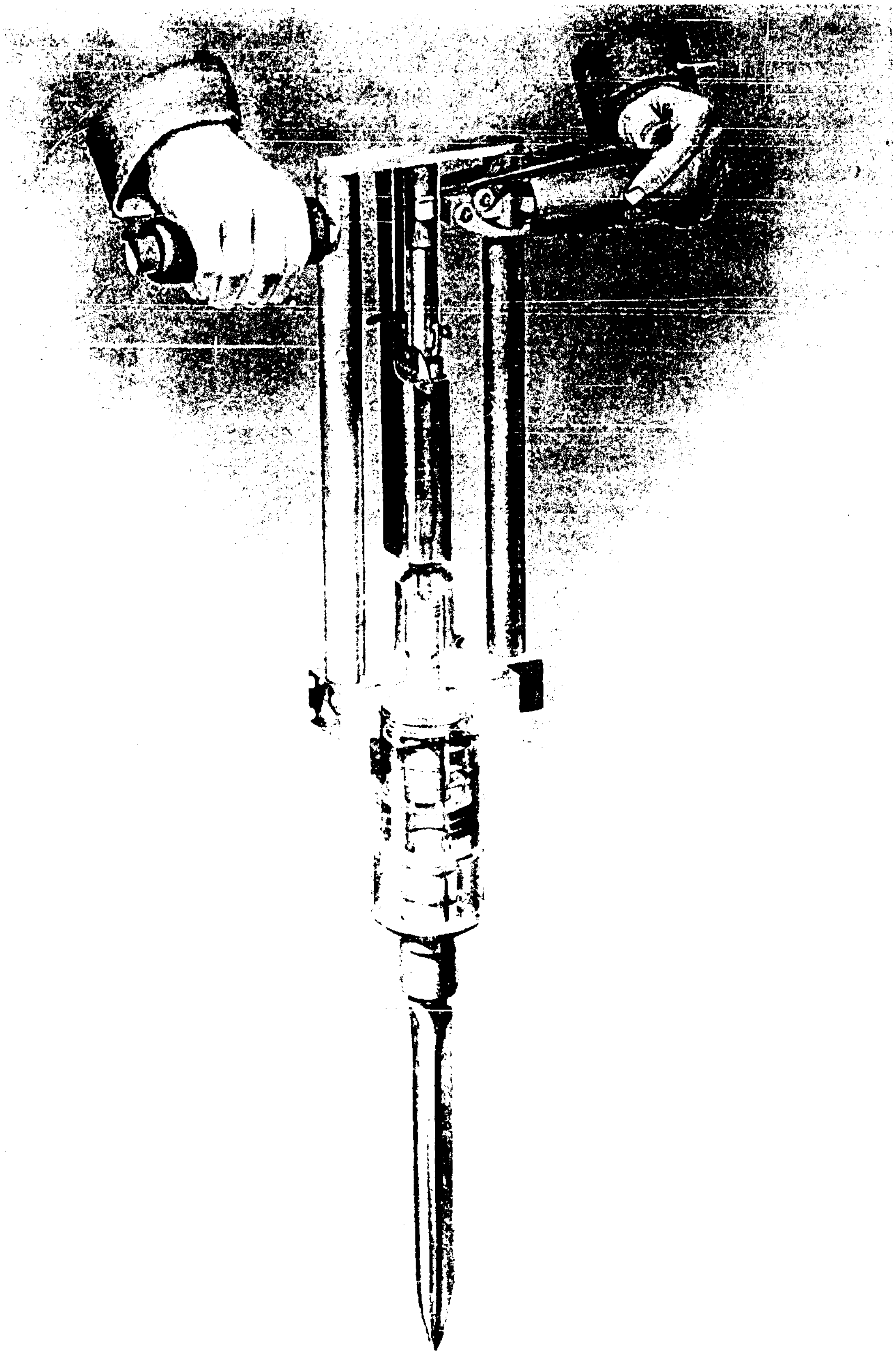
During this report period formal approval of the Second Quarterly Progress Report, AAI Engineering Report 2843A, was received and distribution was made per list supplied by USAELRDL. The Design Plan, AAI Engineering Report 2937, was prepared and delivered. The Visualization Data was prepared and delivered. In addition, two monthly progress reports have been prepared and delivered. These were AAI Engineering Report 2792D covering the period 1 January through 20 January 1963 and AAI Engineering Report 2792E covering the period 21 January through 20 February 1963.

IV. FACTUAL DATA

During the third quarterly report period a comprehensive experimental program was conducted to verify earlier design calculations. Results of this work were incorporated into the design for the device. Experimental investigations were conducted employing a preliminary device to establish maximum limits of important design parameters. Such basic data included: the accurate correlation between charge weight, internal volumes, piston weight and piston velocity. Having this data permitted a further correlation with recoil momentum and the effects on an operator emplacing ground stakes with various projectile energies. After the maximum recoil that an operator can comfortably tolerate had been established, various piston and impact hammer configurations could be investigated based on this momentum. Further, experimental penetration data will be obtained employing the model presently being fabricated to assure accurate relations between penetration, soil types and emplacement energy.

By conducting these tests in an orderly fashion as outlined in the Design Plan, and obtaining the desired data. an optimized design can be obtained to achieve the required performance with maximum reliability and minimum expenditure.

The design of the impact hammer as now developed is shown in Figure 1 . The impact hammer is designed to emplace and retrieve stakes by using the effect of multiple dynamic loadings. Multiple dynamic loadings are employed because a single dynamic load would transmit intolerably large impulse to the operator.



GROUND STAKE EMPLACEMENT-RETRIEVING DEVICE

Figure 1

Basic operation of the impact hammer is as follows:

Upon ignition of the propellant charge (standard M6 grenade cartridge) the energy of the propellant gases acts upon the head of the piston. At the start of its motion, the piston begins to compress the large diameter helical compression spring. After the piston has traveled approximately one inch (within which distance it achieves its maximum velocity) it uncovers the exhaust ports, allowing the escape of the propellant gases. After traveling an additional .5 inch, the piston impacts the stake (or retrieve bar in the retrieve mode), transferring much of its energy to the stake.

At this point the helical compression spring is compressed and the kinetic energy of the piston is nearly zero. The spring then returns the piston to its initial starting point. Once the semiautomatic cal. 30 carbine mechanism, which has extracted the spent case and cycled the bolt, is actuated by the operator, the second propellant charge is ignited and the operating cycle is repeated.

A. Recoil Effects

Driving a piston forward and bringing it to rest by an external force, the ground stake imparts an equal and opposite momentum to the ballistic impact hammer. The resultant momentum must be within the tolerable limitation of a human operator to safely overcome and arrest the recoil motion. This is the design criteria that determines the maximum output of hand-held ballistic devices.

Several cartridges with various charge weights were fired in the preliminary test device to measure the piston velocity as a function of charge weight. Repeated firings were made until the piston velocity, and hence its momentum, were accurately established.

These incremented cartridges were then fired against rigid mediums simulating the effect of encountering solid rock. A comfortable maximum recoil was established with a 10-grain charge corresponding to 12 lb.-sec. of momentum. Somewhat beyond this point the operator could not restrain the impact hammer from leaving the stake head and felt that a conscious effort was required.

From these findings any ballistic system designed for this task should not exceed 12 lb.-sec. recoil momentum. This establishes the piston weight for any desired piston velocity.

B. Emplacing Energy

The preliminary test device employed a 5 lb. piston with 77 ft./sec. impact velocity. This corresponds to 12 lb.-sec. momentum and 460 ft.-lbs. of energy. Since the recoil momentum can not be increased, the obvious step to achieve more penetration is to increase the energy by reducing the piston mass to maintain no more than 12 lb.-sec. momentum.

Penetration data was taken with Ground Stake GP-112/G in frozen river ice and frozen soil and is presented in Figure 2. Eleven rounds with 460 ft.-lbs./rnd. were used for the ice and five rounds of the same energy for emplacement in frozen ground.

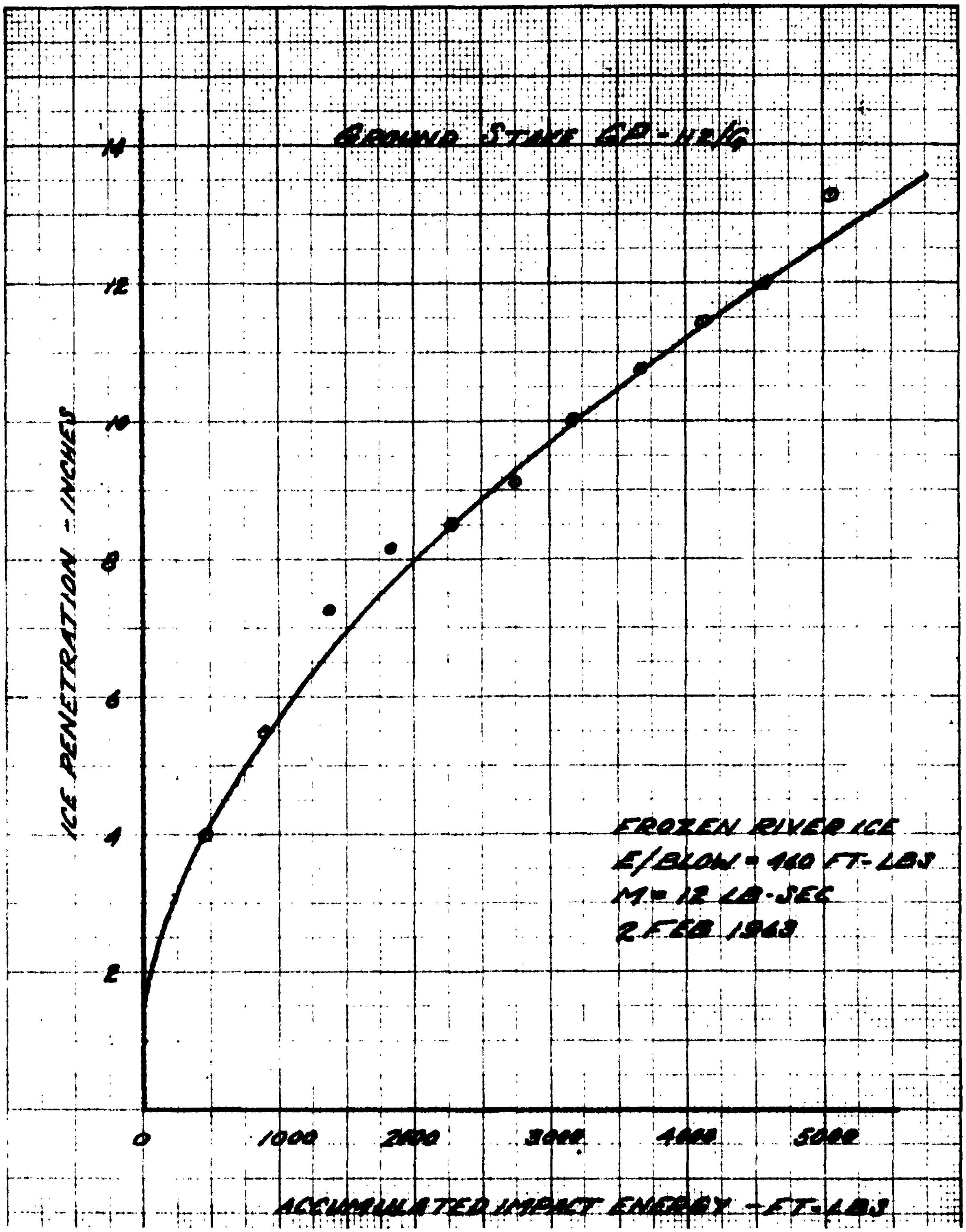

GROUND STAKE GP-12/G
ICE PENETRATION - INCHES
14
12
10
8
6
4
2
0
1000
2000
3000
4000
5000
ACCUMULATED IMPACT ENERGY - FT-LBS
FROZEN RIVER ICE
E/BLOW = 440 FT-LBS
M = 12 LB-SEC
2 FEB 1963

Figure 2

The number of rounds for emplacement is expected to decrease linearly with increase in piston energy. For this reason, a 3-pound piston, which is considered the smallest practical size, has been selected increasing the energy to approximately 800 ft-lbs., theoretically halving the required rounds with the 5-pound piston.

C. Retrieving Energy

Approximately 3 rounds or approximately 2000 ft-lbs. of hammer energy, were used to extract the Ground Stake GP-112/G from a permafrost soil sample. Retrieval was accomplished by pulling the stake with the recoil energy of the impact hammer mockup. A collar attached to the hammer was pin connected to the stake. Recoil was maximized by emplacing the hammer over a flat end bar which rested directly on the permafrost. The bar prevented the piston from moving and in effect drove the hammer upward extracting the stake. Figure 3 schematically illustrates the sequential events of extraction of this mockup retrieve device. Note that the pin connection assures that the stake can never leave the hammer from over-extraction forces.

D. Interior Ballistics

1. Analytical Development

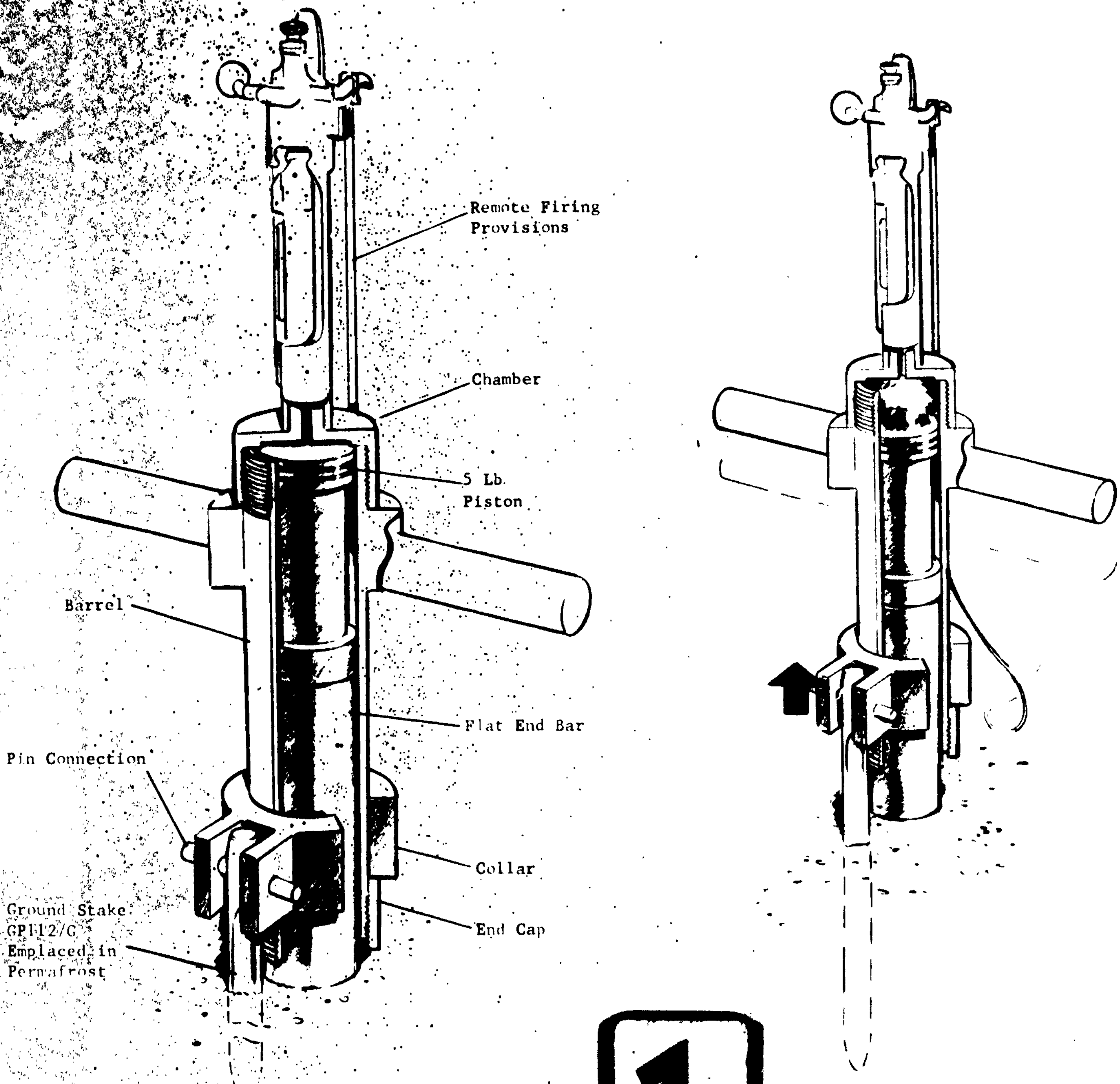
Certain desired design parameters for the impact hammer will prescribe the associated interior ballistics. These parameters include:

a. Hammer energy required

b. Maximum allowable pressure

c. Maximum desired exhaust pressure

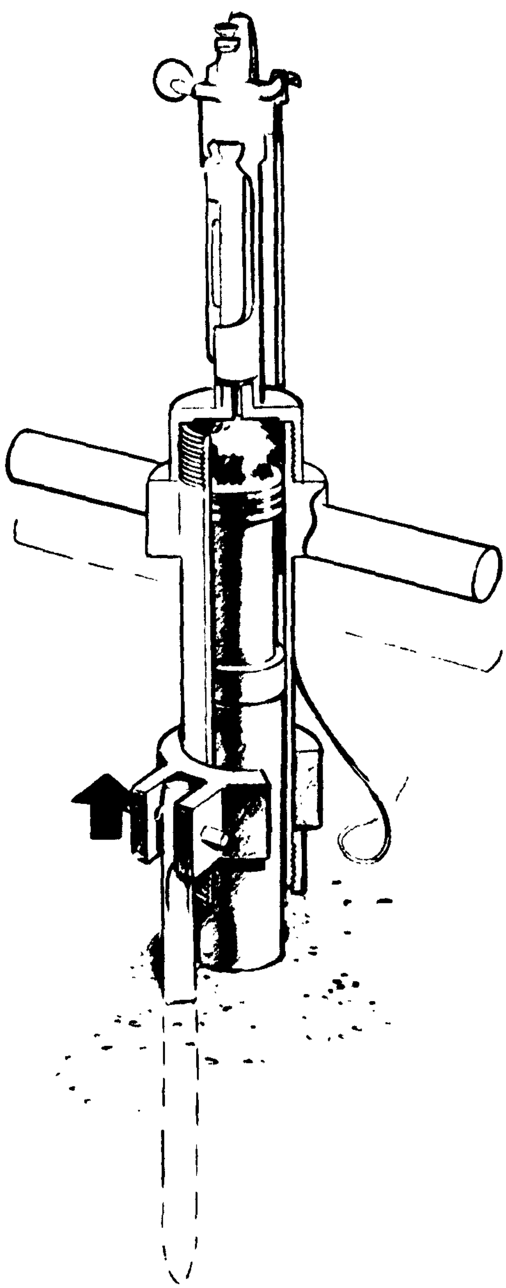
d. Piston area



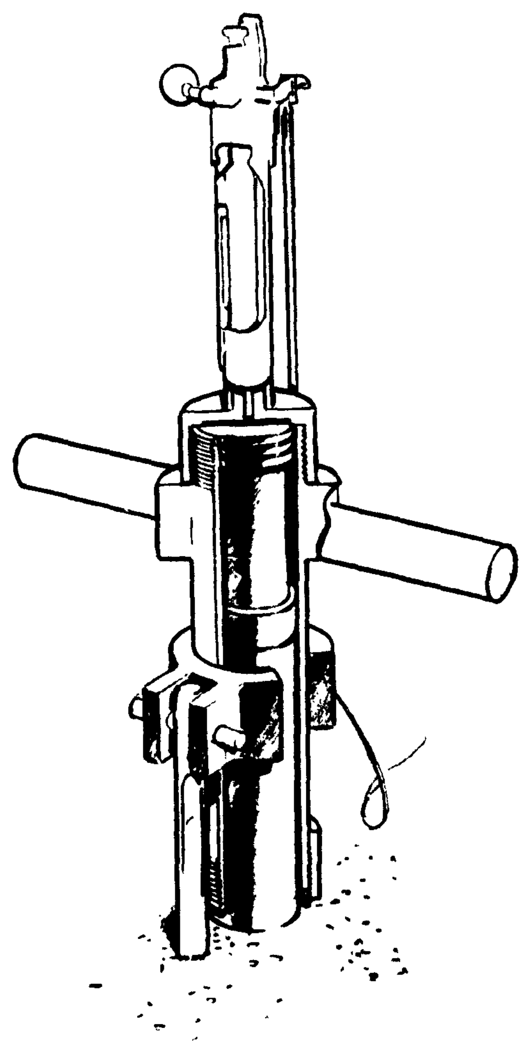
BEFORE FIRING

RECOIL EXTRACTION

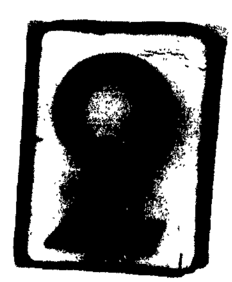
1



RECOIL EXTRACTION



AFTER RECOIL



EXTRACTION SEQUENCE
(MOCKUP RETRIEVE DEVICE ILLUSTRATED)
Figure 3

The piston energy is derived thermodynamically from extremely hot propellant gases. The magnitude of pressure created in a fixed volume by a given propellant is described by the propellant impetus as

$$P_1 = \frac{12\ CF}{V_1} \qquad (1)$$

where:

C	= charge weight	lbs.
F	= propellant impetus	ft-lb/lb.
P_1	= pressure	lb/in^2
V_1	= fixed volume	$in.^3$

As the gas expands from one state to another, assuming an adiabatic relation, the work done is

$$W = \frac{P_1V_1 - PV}{12(\gamma - 1)} \qquad (2)$$

where:

P	= pressure at any volume, V	lbs/in^2
V	= second state of expansion	in^3
P_1	= initial pressure	lb/in^2
V_1	= initial volume	in^3
W	= work output	ft-lbs.
γ	= ratio of specific heats of propellant gases	

From the adiabatic relation

$$P = P_1 \left(\frac{V_1}{V}\right)^{\gamma} \qquad (3)$$

Combining equations (2) and (3)

$$W = \frac{P_1V_1\left[1 - \left(\frac{V_1}{V}\right)^{\gamma - 1}\right]}{\gamma - 1} \tag{4}$$

Substituting CF for P_1V_1 and describing the work done as the kinetic energy of the hammer

$$1/2\ mv^2 = \frac{CF\left[1 - \left(\frac{V_1}{V}\right)^{\gamma - 1}\right]}{\gamma - 1} \tag{5}$$

where: m = mass of hammer slugs

v = velocity of hammer ft/sec.

One final relation is the volume expansion. Part is due to the piston movement and part due to the hammer itself as it recoils.

$$V = V_1 + A(X + X_H) \tag{6}$$

where: A = piston area $in.^2$

X = piston movement in.

X_H = hammer movement in.

From the conservation of momentum the velocity of the hammer will be related to the piston by

$$m_H\ v_H = m_P\ v_P \tag{7}$$

$$\therefore\ X_H = \frac{m_P}{m_H}\ X$$

Then equation (6) can be written

$$V = V_1 + A(X + \frac{m_P}{m_H} X)$$

$$V = V_1 + (1 + \frac{m_P}{m_H}) AX \qquad (8)$$

Equations (5) and (8) relate the derived piston energy from a given charge weight at a given piston stroke, x, measured relative to a fixed axis. The following table is prepared assuming a hammer piston with a 1-inch drive diameter and an initial volume $V_1 = .5 \text{ in}^3$. Using the standard carbine grenade cartridge as the power source provides 20 grains of propellant. The impetus is estimated as F = 300,000 ft-lb/lb. corresponding to a peak pressure of 20,600 psi in the locked shut condition. This pressure seldom occurs as the piston usually moves creating a larger volume when all the propellant is burned. None of the energy is lost by assuming all burning occurs in the initial volume but the peak pressure is something less than indicated at $x = o$. Explosive ordnance experience shows that this type propellant usually burns in .2 to .3 millisecond and from the pressure-travel curve, a fair estimate of the peak pressure can be obtained on the adiabatic expansion curve.

Increments of time are obtained over small travel increments using the mean velocity over that increment. The hammer weighs 22 pounds and the piston 3 pounds.

① x (in.)	② 1.138AX (in^3)	③ V (in^3)	④ V/V_1	⑤ $④^{.25}$	⑥ 1/⑤	⑦ 1–⑥	⑧ $\frac{CF}{\gamma-1}$ ⑦ (ft-lb)	⑨ v (ft/sec)	⑩ $④^{1.25}$	⑪ P (psi)	⑫ Δt (millisec)	⑬ t (millisec)
										20,600	.307	
.1	.0894	.589	1.178	1.042	.960	.040	137	54.2	1.227	16,800		.307
											.130	
.2	.1788	.679	1.358	1.080	.926	.074	254	73.8	1.465	14,000		.437
											.306	
.4	.3575	.858	1.716	1.144	.874	.126	432	96.2	1.962	10,500		.743
											.255	
.6	.5360	1.036	2.072	1.200	.833	.167	572	111	2.485	8,300		.998
											.224	
.8	.7150	1.215	2.430	1.248	.801	.199	682	121	3.035	6,800		1.222
											.210	
1.0	.8940	1.340	2.680	1.280	.781	.219	750	127	3.430	6,000		1.432

TABLE OF CALCULATIONS

This data is plotted in Figure 4 with the estimated time-pressure trace. The travel is relative to the hammer equal to $(1+\frac{m_P}{m_H})X$ and determines the actual distance required by the piston.

Corresponding to 127 ft/sec, the 3-pound piston momentum is

$$I = mv$$
$$= \frac{3}{32.2} \times 127$$
$$= 11.8 \text{ lb-sec.}$$

The energy in one blow

$$E_k = 1/2 \; mv^2$$
$$= 1/2 \; \frac{3}{32.2} \times 127^2$$
$$= 750 \text{ ft-lbs.}$$

2. Experimental Results

Reference to the Table of Calculations (page 14) indicates that the desired piston velocity of 127 ft/sec. is attained with a one-inch travel. The maximum gas pressure is approximately 16,500 psi. In order to verify these analytical results a test program was conducted. Two components were fabricated with the design geometry (from an interior ballistics standpoint) of the chamber and the three-pound piston. Sketches of these components may be found on page 17 . The Caliber 30 carbine mechanism was modified and attached to the chamber by three mounting bolts.

The first tests were made with the gas pressure acting on the piston for a stroke of 2.31 inches. This produced an excessive piston

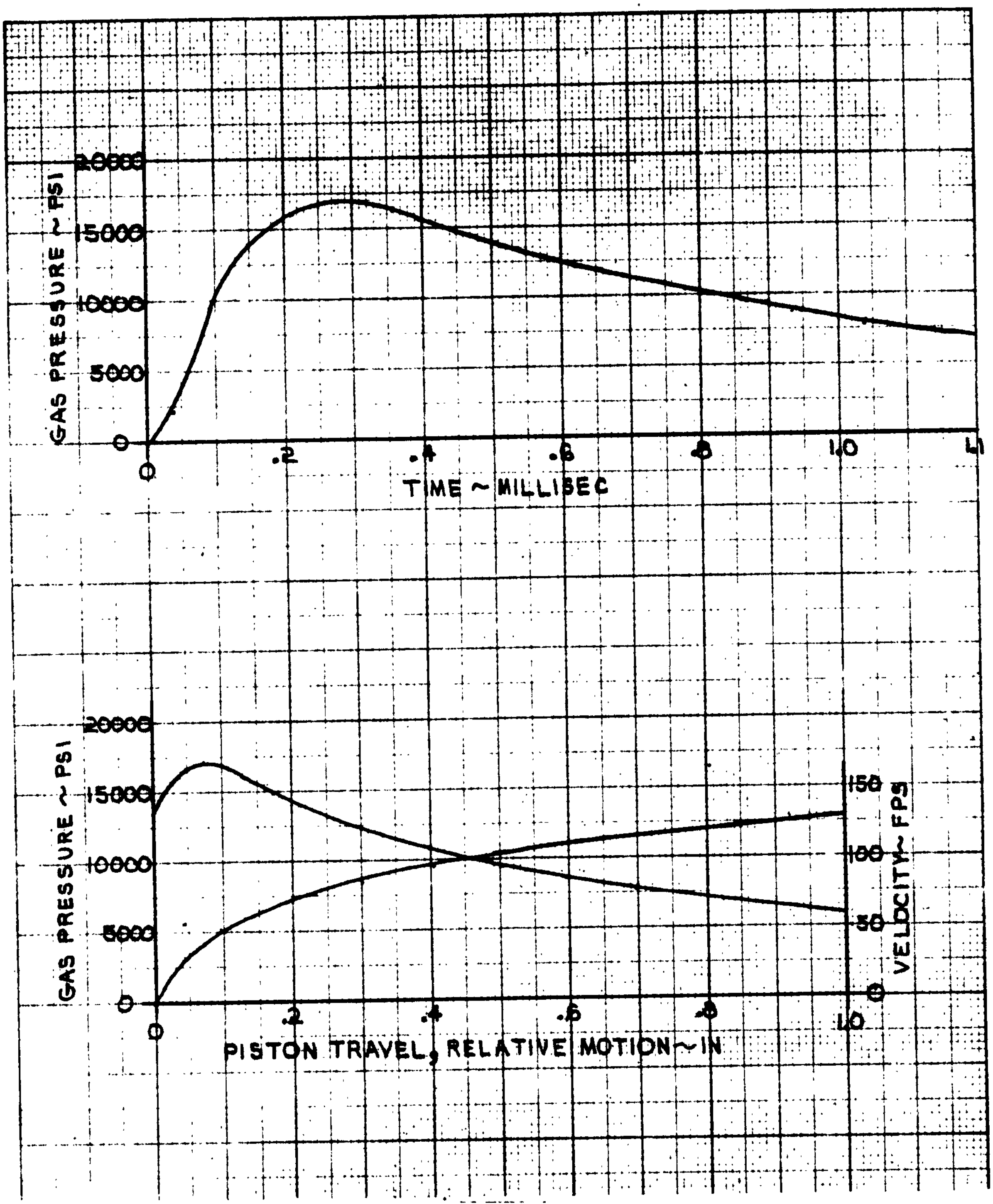
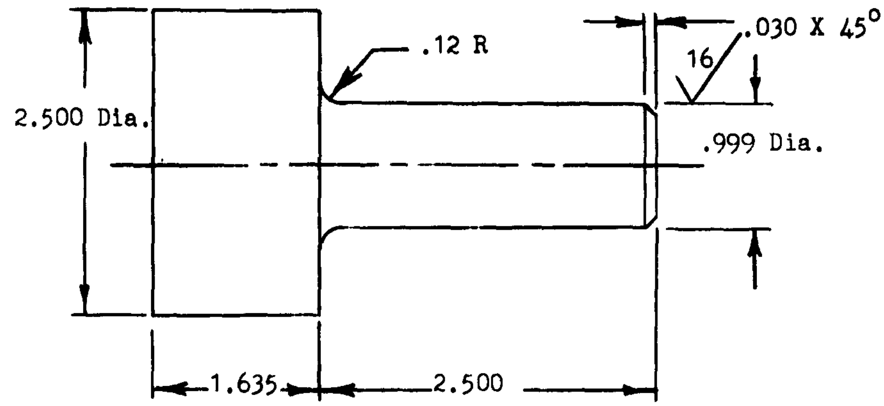
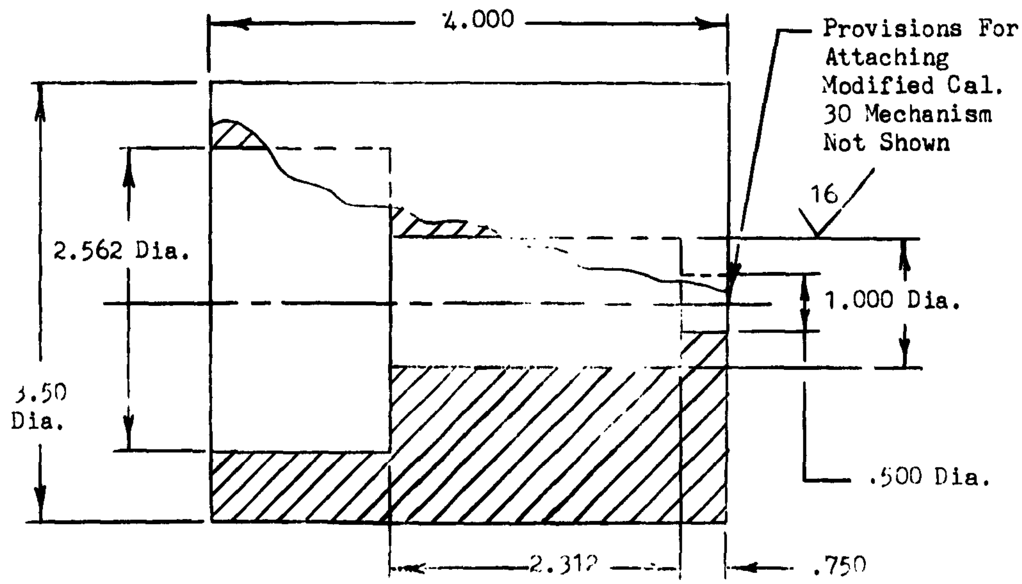

GAS PRESSURE ~ PSI
20000
15000
10000
5000
0
0
.2
.4
.6
.8
1.0
1.1
TIME ~ MILLISEC
GAS PRESSURE ~ PSI
20000
15000
10000
5000
0
0
.2
.4
.6
.8
1.0
150
100
50
0
VELOCITY ~ FPS
PISTON TRAVEL, RELATIVE MOTION ~ IN

Figure 4



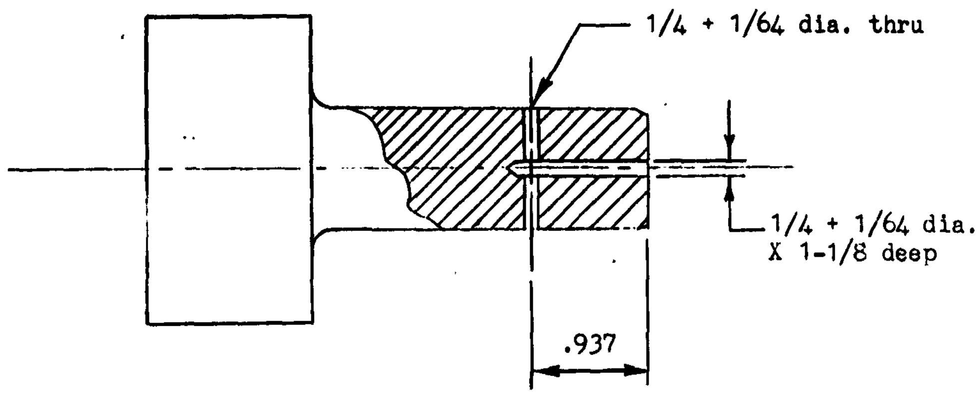
PISTON MOCKUP
(Wt. 3 lb.)



CHAMBER MOCKUP

Figure 5

velocity, as expected. Ports were drilled into the piston (see below) so that the gas pressure would exhaust after .94 inches of stroke. This re-



PORT LOCATION, PISTON MOCKUP

sulted in a piston velocity of 121 ft/sec, which is sufficiently close to the design velocity of 127 ft/sec.

After the design velocity had been attained, pressure-time data were taken for the device. The results are shown on page 19 . The maximum pressure was found to be 17,000 psi, compared to an expected maximum of 16,800 psi (see page 16). It should be pointed out that while the analytical and experimental maximum pressures agree, the shapes of the analytical and the experimental pressure-time curves appear to be different. This is because the analytical pressure-time curve was assumed at the point of ignition. The experimental results, however, incorporated a time lag while the piston moved before uncovering the pressure tap. From time = 0 to time = .3 milliseconds the piston, because of its inertia, began slowly

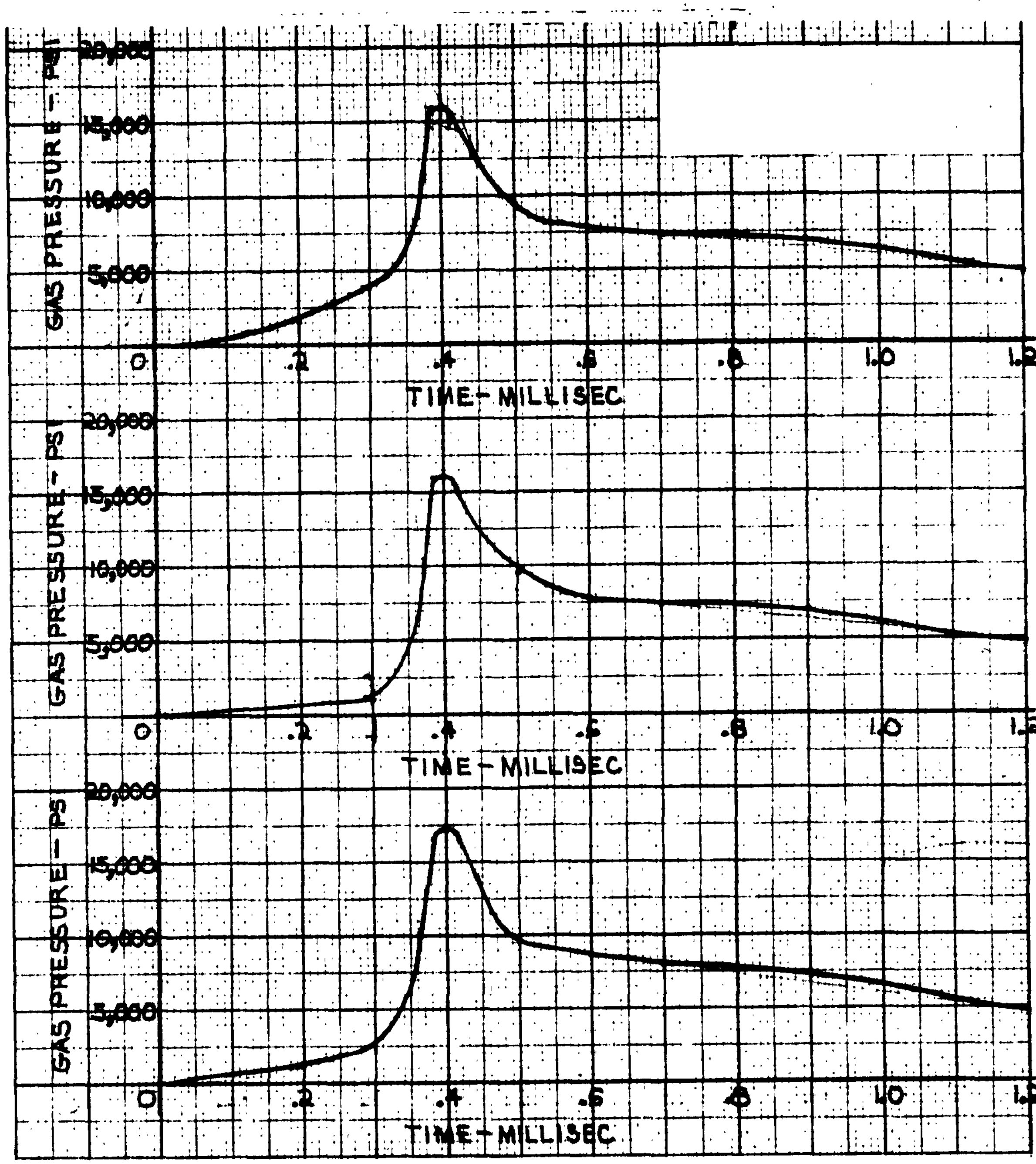

GAS PRESSURE - PSI
20,000
15,000
10,000
5,000
0
.2
.4
.6
.8
1.0
1.2
TIME - MILLISEC
GAS PRESSURE - PSI
20,000
15,000
10,000
5,000
0
.2
.4
.6
.8
1.0
1.2
TIME - MILLISEC
GAS PRESSURE - PSI
20,000
15,000
10,000
5,000
0
.2
.4
.6
.8
1.0
1.2
TIME - MILLISEC

Figure 6

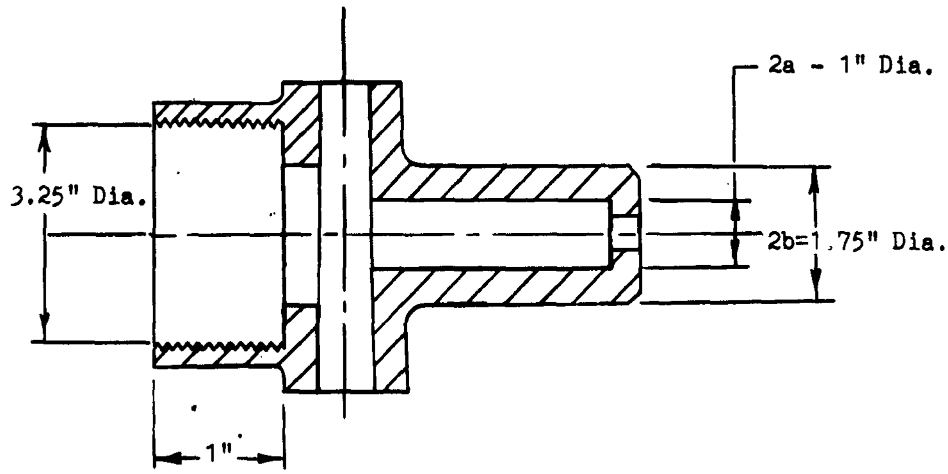
uncovering the pressure tap. As it uncovered more and more of the tap, increased amounts of entering gases provided an indication of a rapidly rising pressure coinciding with the predicted p-t curve.

E. Stress Analysis

The following analysis substantiates sufficient factors of safety on all components in stressed areas. Each part is presented with the type material and material properties. Endurance limits in shear and bending are assumed as a fraction of the ultimate stress.[1] The endurance limit in shear is assumed as 0.53 the endurance limit in bending, and the endurance limit in direct stress is assumed as .85 the endurance limit in bending. When the endurance limit in bending is not given, it is assumed as 0.44 the ultimate stress.

The design general arrangement is shown in Figure 7.

1. Chamber (Type 416 C-R Steel)



(1) Vidosic, J. P., Machine Design Projects, Donald Press Co., New York, 1957.

As indicated in the Interior Ballistics section, a maximum pressure of 20,600 lb/in^2 can be expected in the locked shut condition. Assuming this pressure the maximum stress is

$$S = P\left(\frac{b^2 + a^2}{b^2 - a^2}\right)$$

$$= 20{,}600\left(\frac{.875^2 + .500^2}{.875^2 - .500^2}\right)$$

$$S = 38{,}100 \text{ lb/in}^2$$

$$\text{Endurance Limit} = .85 \times .44 \times S_u$$

$$= .85 \times .44 \times 185{,}000$$

$$= 69{,}200 \text{ lb/in}^2$$

The factor of safety for infinite life based on the endurance limit is:

$$F.S. = \frac{S_e}{S}$$

$$= \frac{69200}{38100}$$

$$F.S. = 1.8$$

For the threaded section,

$$\text{Shear Area } A_s = \pi DL\,(.5) = \pi \times 3.25 \times 1 \times .5$$

$$A_s = 5.1 \text{ in.}^2$$

$$\text{Maximum force} = F_1 = P_1 \times \frac{\pi}{4}(1)^2 = 20{,}600 \times .784$$

$$F_1 = 16{,}300 \text{ lbs.}$$

$$\text{Shear Stress} = S_s = \frac{F_1}{A_s} = \frac{16{,}300}{5.1}$$

$$S_s = 3{,}200 \text{ lb/in}^2$$

2. Piston (tool steel)

The ability of the piston to withstand the most severe loading expected will be analyzed. This would occur if Ground Stake GP113/G encountered an impenetrable medium during emplacement. Under these circumstances, nearly all of the piston's impact energy would be absorbed as strain energy.

Assume elastic conditions;

Maximum kinetic energy of the piston = KE_1

$$KE_1 = 1/2(m_p v_p^2) = 1/2 \left(\frac{3}{32.2}\right) 121^2$$

$$KE_1 = 680 \text{ ft-lb.}$$

$$KE_1 = \frac{F_1^2 L}{2AE}$$

where: F_1 = the maximum force developed in the piston

L = stake length

A = stake area

$E = 30 \times 19^6 \text{ lb/in}^2$

$$F_1^2 = KE_1 \left(\frac{2AE}{L} \right)$$

$$= 680 \left(\frac{2(1.1)30 \times 10^6}{4}\right)$$

$$F_1^2 = 1.12 \times 10^{10}$$

$$F_1 = 106{,}000 \text{ lb.}$$

$$S_{max} = \frac{F_1}{A_p} = \frac{106,000}{1.1}$$

$$S_m = 97,000 \text{ lb/in}^2$$

The tensile strength of the tool steel is $S_u = 320,000$ lb/in^2 indicating an endurance limit of

$$\begin{aligned} S_e &= .44 \times .85 \times S_u \\ &= .44 \times .85 \times 320,000 \\ &= 120,000 \text{ lb/in}^2 \end{aligned}$$

$$F.S. = \frac{S_e}{S}$$

$$= \frac{120,000}{97,000}$$

$$F.S. = 1.24$$

Further calculations have shown that if Stake GP-112/G encountered a solid medium during emplacement, the resulting stress level in the piston would be one-half as that above.

3. Exhaust Tubes & Handles

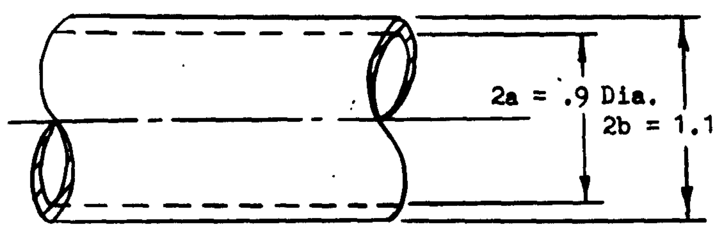
a. Stresses due to gas pressure

Assuming an isothermal expansion into the large low pressure volume,

$$P_2 = \frac{P_1V_1}{V_2} = \frac{20,600 \times .495}{(.495 + 19.82)}$$

$$P_2 = 510 \text{ lb/in}^2$$

$$S_{max} = P_2 \frac{b^2 + a^2}{b^2 - a^2}$$



EXHAUST TUBE

$$S_m = 510 \frac{(.55)^2 + (.45)^2}{(.55)^2 - (.45)^2}$$

$$S_m = 2580 \text{ lb/in}^2$$

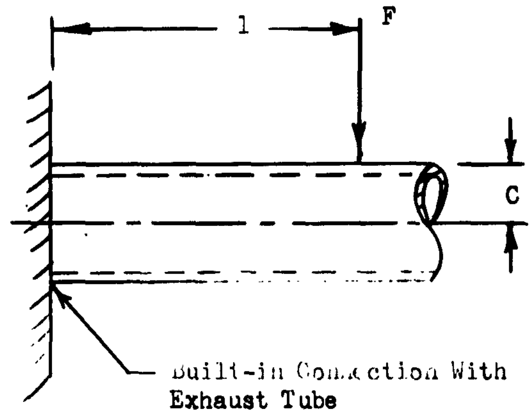
b. **Maximum bending force permitted = F_m**

Since the exhaust handles are subject to severe abuse during transportation and handling, an estimate is made of the maximum load that can be safely withstood.

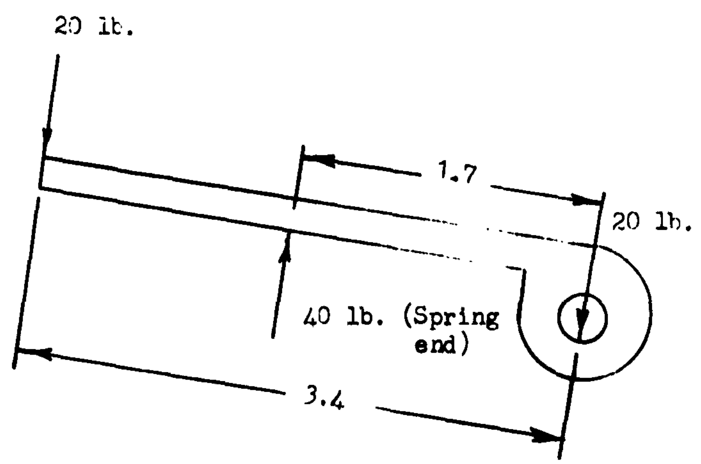
$$S = \frac{F \ell c}{I} = 20{,}000$$

$$F_m = \frac{20{,}000 \times \pi \times .216 \times .08}{3.36}$$

$$F_m = 400 \text{ lb.}$$


1
F
C
Built-in Connection With
Exhaust Tube

4. **Safety**

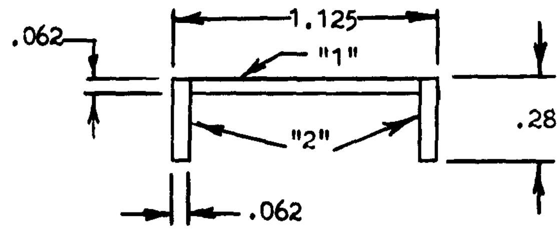

20 lb.
1.7
20 lb.
40 lb. (Spring
end)
3.4

The maximum bending stress will be

$$S = \frac{Mc}{I}$$

Reaction at spring end = $\frac{20 \times 3.4}{1.7} = 40$ lb.

The maximum moment developed = 34 in-lb.



CROSS-SECTION AT LOCATION OF MAXIMUM MOMENT

$$I_1 = \frac{1.125}{12} \times (.062)^3 = 22.3 \times 10^{-6} \quad \text{in.}^4$$

$$I_2 = 2 \times .062\ (.28)^3 = 2.73 \times 10^{-3} \quad \text{in.}^4$$

$$\therefore\ I_T \approx 2.74 \times 10^{-3}$$

$$S_{max} = \frac{Mc}{I} = \frac{34(.14)\ 10^3}{2.74}$$

$$= 1750 \text{ psi}$$

5. Barrel

If the hammer is actuated with the stake not in place, the kinetic energy of the piston must be absorbed by the hammer barrel. In this event, the piston impacts a series of elastomer buffers limiting the impact force in the order of 150,000 lbs. This dynamic force depends on the buffer material and permissible deflection. Experimental verification is necessary to confirm the magnitude of this impact.

A wall thickness of .25 inch has been designed to absorb the severest impact load with the following stress:

$$S = \frac{F}{\pi d t}$$

$$= \frac{150,000}{\pi \times 3 \times .25}$$

$$S = 64,000 \quad lb/in^2$$

After experimentally determining the loads, it may be permissible to decrease the wall thickness to reduce the overall tool weight.

6. Springs

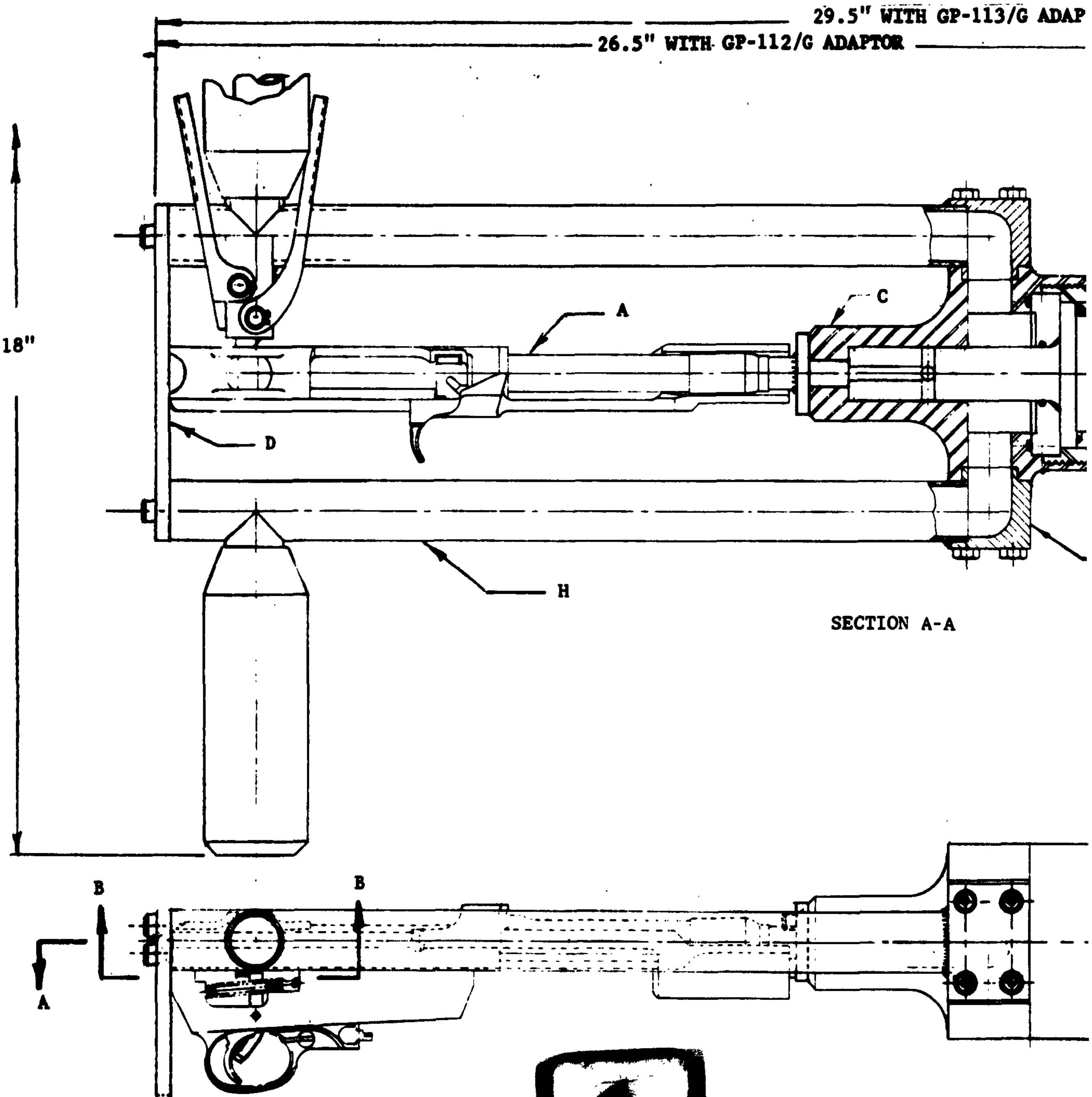
Other areas of significant stress include the piston return spring and the trigger return spring. These springs have been selected according to recommended manufacturer's stress levels.

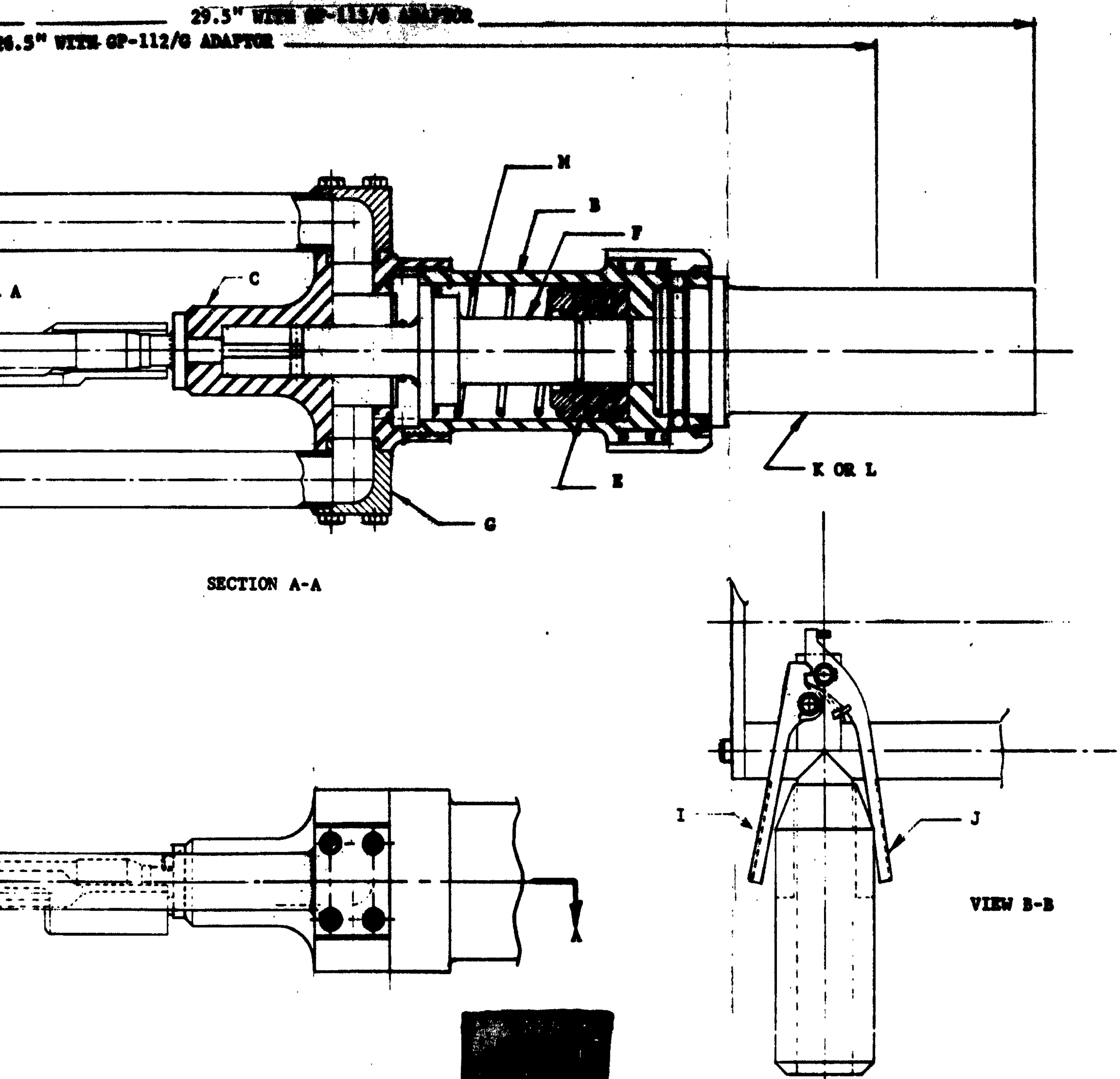
The design limit of NS-355 stainless steel spring wire is 145,000 lb/in^2. This material when used normally in the ballistic impact hammer has been designed to 84,000 lb/in^2. When the accidental firing occurs bottoming the piston against the buffers, the spring stress is 134,000 lb/in^2, still having a safety factor of 1.08. Again this is based on endurance limits.

The function of the trigger spring is to return the trigger after releasing. Its stress level is 138,000 with a safety factor of 1.05.

F. Materials Selection

The materials selected for the major components of the ground stake emplacement-retrieving device are found in the table following. Where


29.5" WITH GP-113/G ADAP
26.5" WITH GP-112/G ADAPTOR
18"
A
C
D
H
SECTION A-A
B
B
A



Stake Emplacement and Retrieving Tool Design Layout
Figure 7

applicable, the selection of materials was based on the results of the stress analysis and the life cycle calculations. It should be pointed out that all materials selected are readily available and that all fabricated metal parts, with the exception of the piston, are of corrosion resistant steel.

Part Letter	Part Name	Material
B	Barrel	Type 416 corrosion resistant steel, heat treated to a tensile strength of 185,000 psi.
C	Chamber	Type 416 corrosion resistant steel, heat treated to a tensile strength of 185,000 psi.
D	Brace	Type 304 corrosion resistant steel.
E	Shocks	Butyl rubber 80-90 Durometer.
F	Piston	"Solar" tool steel, heat treated to a tensile strength of 320,000 psi.
G	Exhaust manifolds	Type 304 corrosion resistant steel.
H	Exhaust pipes - handles	Type 304 corrosion resistant steel tubing.
I	Safety	Type 17-7 PH condition "A" corrosion resistant steel, heat treated to a tensile strength of 220,000 psi.
J	Trigger	Type 17-7 PH condition "A" corrosion resistant steel, heat treated to a tensile strength of 220,000 psi.
K	Large Stake Adaptor	Type 416 corrosion resistant steel, heat treated to a tensile strength of 185,000 psi.
L	Small Stake Adaptor	Type 416 corrosion resistant steel, heat treated to a tensile strength of 185,000 psi.
M	Piston Return Spring	NS355 corrosion resistant steel spring wire.

G. Tool Roll

In addition to the tool and retrieve mechanism, the design of a tool roll to permit shoulder and hand carry has been developed during this quarter. The tool would ordinarily be transported and stored in this roll. The canvas roll is designed to carry four thirty-round clips thus providing for 120 shots. Based upon the energy requirements determined for stake emplacement under worst anticipated conditions, a minimum of 5 stakes could be emplaced with these cartridges. Provision is included in the design to secure each separate tool component to the roll thus preventing collision between parts when the package is subjected to vibration or drop. Pockets are utilized to house the cartridge clips for maximum protection. The carrying straps are designed to permit the roll to be carried on the back or hand carried as illustrated in Figure 8.

H. Weight Analysis

The weights of the major components of the emplacement and retrieving device have been calculated and are tabulated below:

Item	Weight (lb)
Chamber	6.53
Barrel	5.38
Piston	3.00
Caliber 30 rifle mechanism, including full magazine	2.66
Adaptor for Large Stake	2.75
Adaptor for Small Stake	1.45

Item	Weight (lb)
Exhaust pipes	2.36
Exhaust manifolds	1.68
Handles	.98
Tool Roll	3.00
TOTAL	29.79

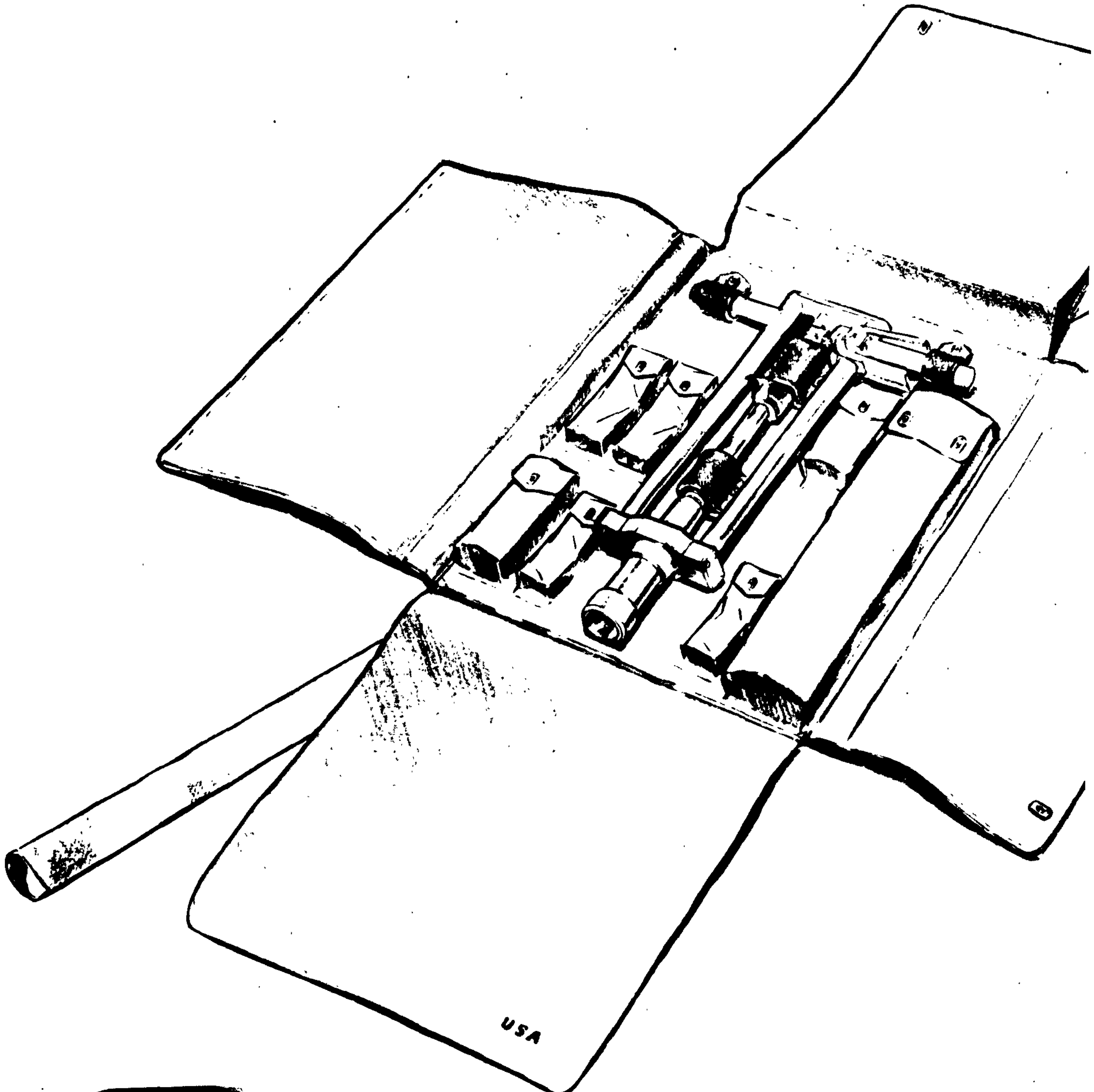
While this weight is above the 25-pound limit, it is felt that future strain-gage testing will dictate weight reductions in certain areas, such as the barrel and chamber. Also it appears reasonable to consider aluminum construction for the two stake adaptors and the complete exhaust system.

I. Drawings Status

The drawings of all the components of the impact hammer which require fabrication have been completed and released to the AAI Machine Shop. Each component has been stress analyzed and toleranced to be consistent with economical fabrication techniques. As minor changes are indicated as the result of fabrication experience, they will be incorporated into the drawings.

J. Manufacturing Status

All components which require fabrication are presently being fabricated in the AAI Machine Shop. As of this date this overall fabrication is 60 percent complete. All components which require fabrication by, or purchase from, outside industry have been ordered.



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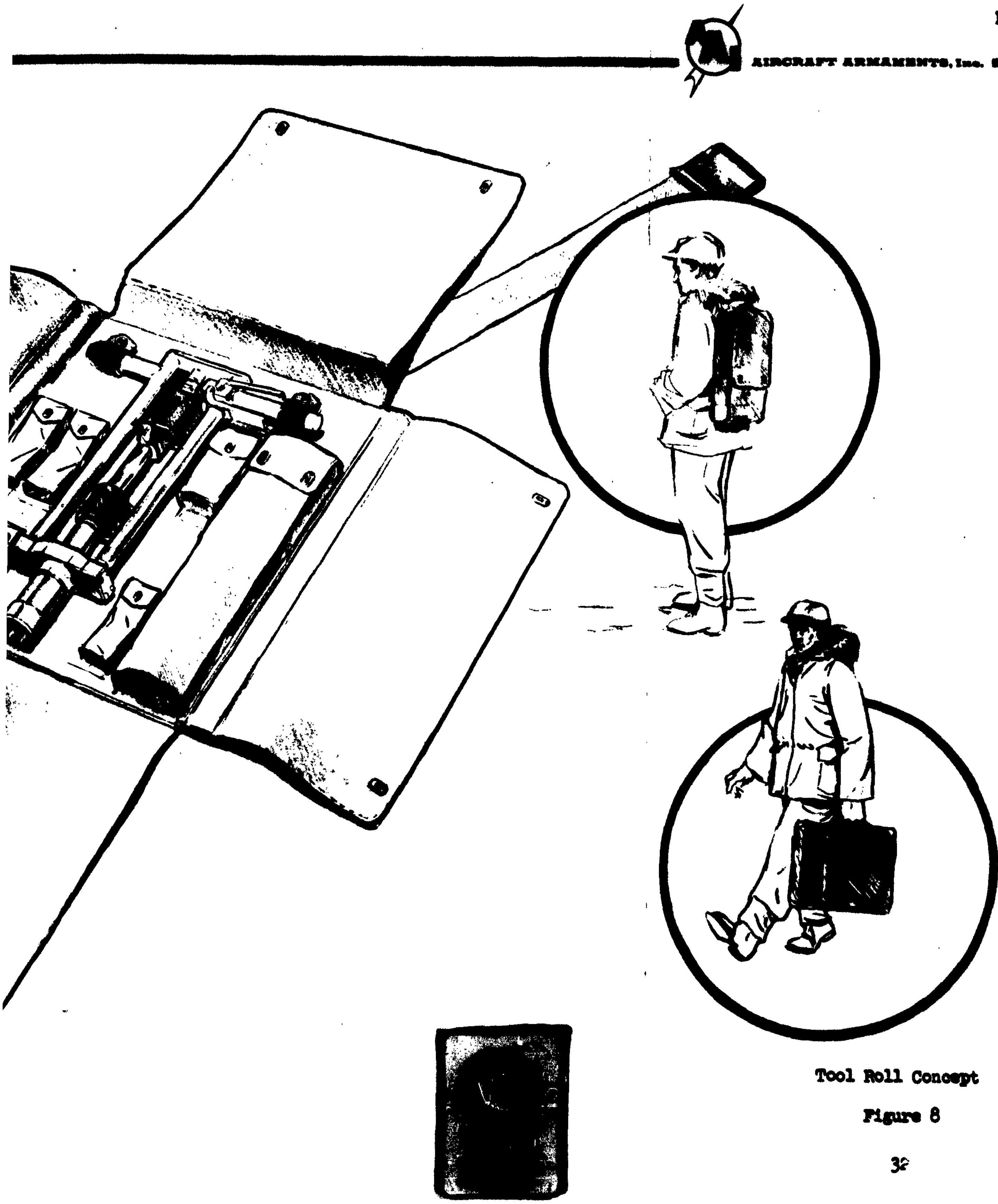

Tool Roll Concept

Figure 8

V. CONCLUSIONS

During this period extensive design analysis, stress analysis, material study and experimentation was conducted and the design for the ballistic impact hammer formulated. Detail parts design including materials selection and stress analysis was accomplished, and following Design Plan approval, drawings were released to permit fabrication of one model.

The experimental work conducted employed a ballistic test model (breadboard) which permitted determination of emplacement energies in various soil types with the applicable Ground Stake. Experimental work has also been performed to evaluate Ground Stake retrieve techniques.

The primary consideration during all design activity has been the ability of the design to satisfy Signal Corps requirements for an emplacement-retrieving device. Parameters not directly related to performance, such as flash, noise, human engineering, safety, logistics, maintenance and simplicity have also been considered at all times.

VI. PROGRAM FOR NEXT INTERVAL

During the next quarterly interval the following items are scheduled to be accomplished:

The first model will be completed and tested according to the Test Plan outlined in the Design Plan. Any changes indicated as desirable as a result of these tests will be incorporated into the design and the remaining two models will be fabricated and tested.

Monthly progress reports will be prepared and delivered according to schedule.

Final copies of the Third Quarterly Progress Report will be distributed upon draft approval.

The following items will be delivered according to schedule:

1. Draft Final Progress Report
2. 3 Devices
3. 3 Instruction Books
4. Completed Manufacturing Drawings

VII. IDENTIFICATION OF KEY TECHNICAL PERSONNEL

Brief resumes of key technical personnel who have contributed to this program are shown on the following pages.

Irwin R. Barr, Vice President - Development

Mr. Barr is an original member of AAI, first serving as Chief Design Engineer, and attaining his present position in 1960. Between 1952 and 1960, he directed the company's ordnance design activities in the capacity of Chief Ordnance Engineer. He has had extensive experience in both ordnance and missile development, being co-inventor of the Viking rocket control system and holding patents on a variety of ordnance items, including a vehicle, turrets, bearings, special fin stabilized ammunition, and several automatic guns. Outstanding examples of ordnance material developed under his direction at AAI include the T175E1 Dual Purpose Machine Gun, a series of turret-type cupolas, and the T92 mm Gun Tank.

Mr. Barr is a graduate in Aeronautical Engineering (1940) from the Casey Jones School of Aeronautics. He worked for two periods, 1940-1944 and 1946-1950, for the Glenn L. Martin Company, primarily in missiles. During the last of these two periods he served in the U. S. Army Air Force, receiving the Army commendation medal for design of aircraft rocket launchers.

Nicholas J. LaCosta, Manager, Explosive Ordnance Department

Education: Casey Jones School of Aeronautics, A.E.

Work History: 1959 to present, Aircraft Armaments, Inc. Manager, Explosive Ordnance Department. Responsible for directing and coordinating all activities of the Explosive Ordnance Department. These activities include the study, design, development, test and manufacture of systems and components involving propellant actuation, interior and exterior ballistics, terminal ballistics, kinematics, gas dynamics and explosive train design, applications and feasibility studies.

1951-1959, Aircraft Armaments, Inc. Project Manager responsible for stabilized ammunition development program, Corporal Warhead development, a number of explosively actuated bomb racks, canopy removers. thrusters and initiators and control systems. Recent responsibilities also include new infantry weapon concepts, recoilless rifle program, special weapons tools and an automatic sequencing device.

1939-1951, The Glenn L. Martin Company Layout Engineer, Group Engineer, Design Specialist.

1938, Bell Aircraft Company. Layout Engineer.

1936-1937, Consolidated Aircraft Company. Draftsman.

Richard G. Strickland, Senior Design Engineer

Education: City College of New York, B.S., 1953
University of Maryland, Graduate Work.

Work History: 1956 to present, Aircraft Armaments, Inc.
Project Manager responsible for design and development of tools and procedures for special weapons handling and disassembly, underwater timing device, and special devices for underwater recovery operations. Design of special tools for remote handling of explosive devices. Survey and study of CAD in all U.S. aircraft and missiles. Experimental study of recoilless rifle internal ballistics; study of recoilless weapon optimization. Spotting round development. Study of missile fuel spark compatibility. Development of timer concepts, applied explosives research. Design and development of underwater guns and ammunition, measurement of effects of underwater explosions. Responsible for development and fabrication of ML25 Demolition Firing Device and associated test equipment. Study and development of wide area mine fuze. Design and development of Ground Stake emplacing device and earth anchor concepts.

1954-1956, U. S. Navy Special Weapons Disposal School. Special Weapons Instructor. Nuclear Physics, Health Physics, Weapon Principles, Weapon Configuration and Weapon Effects.

1953-1954, U. S. Navy
Special Weapons Disposal Officer. Field test of special weapons. Attended Nuclear Weapons Training Schools at Field Command AFSWP, Picatinny Arsenal and Lowry Air Force Base.

Theodore G. Stastny, Senior Engineer

Education:	Georgia Institute of Technology, BME, 1958 Georgia Institute of Technology, MME, 1959
Work History:	1958 to present, Aircraft Armaments, Inc. Responsible for analytical and experimental evaluations of the internal ballistics of recoilless rifles and both hybrid and solid-fuel rockets. Study and design of various cartridge actuating devices employed in ejection mechanisms, parachute deployment, and separation problems. Trajectory, stability and aerodynamic analysis of projectiles and gun boosted rockets. Stress and cycle analysis, including dynamics and kinematics of automatic weapons. Responsible for compiling theories of detonation and explosive phenomena, and fragmentation of bombs and grenades for handbook usage. Designer of "Lo to Hi Pressure Admission Valve" of hot gases (Pat. Pend.). Heat transfer and stress analysis of large, high pressure flat-end chemical reactor. Investigation of underwater ballistics on Mach 0.1 to 0.5 projectiles covering velocity decay, vaporization bubble, stability and nose configurations. Responsible for extensive literature survey and analysis covering techniques of rapid ground anchor emplacement devices including related studies of soil mechanics and soil properties. 1954-1958, Koppers Company, Inc., concurrent with schooling. Design and study of gas cleaning equipment, design of gear teeth for coupling misalignment, studies of sealing ring and piston ring problems.
Technical Society:	Member and past officer of American Rocket Society, Maryland Section.

Arthur C. Powell, Senior Engineer

Education:	Massachusetts Institute of Technology. B.S. in M.E., 1948 U.S. Navy Electronics Technicians Schools 1944-1945
Work History:	1958 to present, Aircraft Armaments, Inc. Senior Engineer. Design and test of a parachute sequencing device, a hydraulic power supply, recoilless weapons, cartridge actuated devices, a remote arming device, escape system components, hydraulic timers, and production tooling.
	1952-1958, Van Zelm Associates, Inc. Design Engineer on a variety of aircraft arresting gears, target drone catapults, and bridle arresting equipment. Field test engineer on aforementioned equipment.
	1949-1952, Friez Instrument Division, Bendix Aviation Corporation. Design and test engineering on numerous weather, recording and electrical instruments.
	1948-1949, Industrial Research Laboratories. Design, construction, and test of a filled-bottle goods inspection machine.

Charles H. Gonnerman -- Assistant Engineer

Education: Drexel Institute of Technology
B. S. in M. E., 1962

Work History: 1962 to present, Aircraft Armaments, Inc.
Assistant Engineer

Layout and detail design, stress analysis, and test work concerned with the Chemical Corp's Chemical Reactor, the Signal Corps Stake Emplacement and Retrieving Device, and their related components. Designed two machines which enabled the study of motion effects on the performance of the Chemical Reactor. Layout design, stress analysis, and weight analysis of projectiles and various weapon components for proposal usage.

1958 - 1962: Chemical Research & Development Laboratories, Army Chemical Center; concurrent with schooling.

Study, design, and test of explosive disseminating devices and related components; trajectory and aerodynamic analysis of projectiles.

VIII. DISTRIBUTION LIST

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Cockeysville, Maryland

RAPID EMPLACING & RETRIEVING DEVICE FOR GROUND STAKES GP-112/G AND GP-113/G
by R. G. Strickland, T. G. Stastny, C. Gonnerman
Third Quarterly Report, 1 January to 31 March 1963
44 pages, 8 figures, 1 reference
Unclassified Report

The Third Quarterly Progress Report discusses detail design, stress, material selection and energy requirements for the ballistic impact hammer. The retrieving of stakes after they have been emplaced is discussed and related experimentation is reported. The design of a tool roll for storage and transportation is also described.

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